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# Preliminary Study of a Novel Compact R718 Water Chiller with Integration of a Single Stage Centrifugal Compressor and Two-Phase Ejectors

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## ABSTRACT

A novel compact R718 water chiller with integration of a centrifugal compressor stage and two phase ejectors as a second stage compression device is proposed. The paper describes the investigations of this R718 refrigerating cycle. The limitations of the R718 centrifugal compressor stage pressure ratio are discussed, and possibilities for introduction of two-phase ejectors in the R718 refrigerating systems with direct evaporation and condensation are analyzed. The complex thermal and flow phenomena connected with additional compression, desuperheating and condensation inside the two-phase ejector flow field are investigated and performance characteristics are estimated. The thermodynamic processes are shown in pressure – enthalpy and temperature – entropy diagrams. Based on the described thermodynamic model, a computational procedure was generated to evaluate the performance characteristics of this R718 refrigerating unit. The implementation of the two-phase ejector in the R718 centrifugal compressor refrigerating systems causes simplification of the R718 unit, and reduction of their size and cost. The energy efficiency of these refrigerating systems may also be increased.

## 1. INTRODUCTION

The global environmental problems connected with ozone depletion and global warming have led to renewed interest in natural refrigerants (ammonia, carbon dioxide, water, air and hydro-carbons such as propane, butane). From numerous perspectives, water is the ultimate choice of refrigerant (Madsboll, 2011). The idea of water turbocompressor refrigerating systems started at the end of 1980s and 1990s (Šarevski, 1996; Koren and Ophir, 1996; Madsboll, 1996; Paul, 2007). By the end of 1990s, R718 centrifugal turbo water chiller had already been developed and manufactured by Institut für Luft and Kältetechnik, Dresden, Germany (Albring and Honke, 2011). Development of an axial turbocompressor for commercial chillers with water as refrigerant by Danish Technological Institute is presented by Madsboll (2011). Implementation of three port condensing wave rotor in R718 centrifugal turbocompressor refrigerating cycle is given by Kharazi *et al*, (2006).

Among the technical and environmental advantages of the water systems, the direct evaporation and condensation are additional advantages for the R718 centrifugal compressor refrigerating systems / heat pumps concerning COP increment. Degradation of the heat in the evaporator and in the condenser causes decreasing of the conventional system COP (for example conventional water chillers for air conditioning applications and water-water heat pumps). The influence of the refrigerant thermodynamic properties on some refrigerating turbocompressor characteristics are presented by Šarevski (1996). The characteristics of steam turbocompressors applied in thermotransforming systems are given by Muller (2001); Šarevski and Šarevski (2011, 2012d).

The theory of gas and steam ejectors is given in the fundamental publications (for example: Abramovic, (1969), Power, **(**1993)). Investigations of the ejectors and ejector refrigerating systems working with various refrigerants are presented by Eames *et al*, (1995); Elbel and Hrnjak, (2008). The optimization of the ejector flow field depends on refrigerating system operating conditions. Theoretical and experimental investigations of the two phase flow ejectors applied in the compressor refrigerating systems as devices for reduction of the throttling losses, or for second step compression in the refrigerating cycles are presented by Šarevski *et al*, (2005); Bergander *et al*, (2008); Butrymowicz, (2003); Elbel and Hrnjak, (2008); Smierciew *et al*, (2011); Banasiak *et al*, (2011). Calculation and analysis of sound velocity in vapor liquid two phase flow, as well as theoretical and experimental investigations of transonic flow phenomena in two phase ejectors are given by Wang and Zhang, (2011); Karwacki *et al*, (2011).

The implementation of the two-phase ejector in the R718 centrifugal compressor refrigerating systems causes simplification of the R718 unit, and reduction of their size and cost. The purpose of this work is to present a novel compact R718 water chiller with integration of a centrifugal compressor stage and two phase ejectors as a second stage compression device, to estimate the performance characteristics of the water chiller and to expose the possibilities for application of the unit of R718 single stage centrifugal compressor / two phase ejector working with water vapor, as natural refrigerant, in the refrigerating / heat pump systems.

## 2. MODEL OF R718 CENTRIFUGAL COMPRESSOR

Centrifugal compressor stage pressure ratio (Π) and peripheral Mach number ($M_u$) are

$$\Pi = (1+\psi(\kappa-1)M_u^2/\eta_p)^{\sigma} \quad (1)$$

$$M_u = u_2/a_0; \quad a_0 = (\kappa\xi_0 RT_0)^{1/2}; \quad \xi_0 = p_0 v_0/(RT_0); \qquad R = R_\mu/\mu; \quad \sigma = (\kappa/(\kappa\text{-}1))\eta_p \quad (2)$$

Effective work coefficient and effective work are

$$\psi = \eta_h\psi_t; \quad l_e = \psi u_2^2; \quad \eta_h \approx \eta_p; \quad \psi_t = k_1 - k_2\varphi_2 ctg\beta_2; \quad \psi_t = \mu_s\psi_\infty; \quad \psi_\infty = 1-\varphi_2 ctg\beta_2 \quad (3)$$

Centrifugal compressor impeller outlet flow rate coefficient is

$$\varphi_2 = V_2/(A_2u_2) = V/(k_{\upsilon_2}A_2u_2) = c_{m2}/u_2; \qquad k_{v2} = V/V_2 = v_0/v_2 \quad (4)$$

Centrifugal stage boundary conditions are connected with: peripheral speed limiting criterion by the impeller strength properties, which is significant for working media with small molecular mass, and with Mach number limiting criterion, significant for working media with large molecular mass.

Water is a medium with small molecular mass. Therefore the R718 centrifugal compressor stage can attain relatively low pressure ratio in conditions of usual peripheral speed. The centrifugal stage can attain a high pressure ratio in conditions of high and extremely high peripheral speed $u_2$ = 500–550 (m $s^{-1}$). The limitation of water vapor centrifugal stage pressure ratio is about Π ≈ 3.5 (for limitation of $u_2 \approx 550$ (m $s^{-1}$)), and the limitation of the corresponding temperature lift is about $\Delta t \approx 20^oC$–$22^oC$. Two stage centrifugal compressors are solution for R718 refrigerating / heat pump systems for air conditioning application (Šarevski, 2012b).

The dimensions of the centrifugal compressor mainly depend on the capacity of the refrigerating system $Q_e$, specific volumetric cooling capacity $q_v$ and required pressure ratio Π. The dependence of the impeller diameter $D_2$ and speed of rotation $n$ on $Q_e$, $q_v$ and Π are determined by the following relations:

$$D_2 = \sqrt{V/(\pi\bar{b}_2\tau_2 k_{v2}u_2\varphi_{2d})}; \qquad n = u_2/(\pi D_2) \quad (5)$$

$$V = Q_e/q_v = Q_e c; \quad c = 1/q_v; \quad k_{v2} = (1+\psi r(k-1)M_u^2/\eta_p)^{\sigma-1}; \quad r = 1-(\varphi_2^2+\psi^2)/(2\psi)$$

The deep vacuum operating conditions and low $q_v$ of the R718 causes large dimensions of the R718 centrifugal compressors. Wide range of capacities of R718 centrifugal refrigerating unites (from tens kW up to (1–2) MW) can be obtained (Šarevski, 2012d). R718 centrifugal stage compressor has high peripheral speed, high Mach number and high pressure ratio. Additionally, Reynolds number of water vapor under deep vacuum conditions is low. High superheating at the compressor outlet, caused by high pressure ratio and high value of the water vapor isentropic coefficient, is a huge problem in R718 centrifugal refrigerating systems.

Theoretical and experimental investigations of high pressure ratio and high Mach number centrifugal compressors have led to establish the following main features of the flow in their flow field: separated jet-wake flow and transonic flow phenomena. An analytical prediction of transonic flows in turbomachinery cascades is generally difficult because of the mixed hyperbolic-elliptic character of the problem. One way of solving the transonic problem is by using the time-dependent and finite-volume methods. These methods are applied for 3-D unsteady viscid flow calculations. Numerous CFD methods and applicative software have been developed in the recent years and are applied for prediction of the flow in the turbocompressors.

If the purpose is to obtain high pressure ratio centrifugal stage with high efficiency then the water vapor properties and design operating conditions should be taken in the optimization design procedure of the compressor flow field. The efficiency is estimated to be $\eta_p$=0.72–0.82. The lowest values correspond to small compressors, where the influence of high Mach number and low Reynolds number is strongly express.

Impellers with lower impeller blade outlet angle $\beta_2$ attain higher peripheral Mach number $M_u$ for a given fluid flow Mach number $M_{w1}$ limitation, but these impellers have lower work coefficient $\psi$. Therefore high pressure ratio impellers should be design with high $\beta_2$ ($\beta_2 = 75^o$–$90^o$), although in their flow field separated jet-wake flow appears. These impellers have high strength characteristics and can withstand high and extremely high peripheral speed. Caused by the influence of the impeller slip factor and of flow structure in the blade – to – blade channel, the number of the impeller blades should be high ($z_2$=25–35). If the impeller dimensions are small then two row cascades can be applied ($z_2=2z_1$). Impellers with lower $\overline{b}_2$ ($\overline{b}_2$ =0.02–0.04) attain higher $M_u$ and respectively higher $\Pi$ for a given $M_{w1}$ limitation (Šarevski, 2012d).

## 3. MODEL OF R718 TWO PHASE EJECTOR

Compression in these two phase ejector systems is realized by using motive water with high pressure. Water pressurized by a hydraulic pump as a motive fluid is used in the two phase ejector vacuum pump systems and also in the concentrator thermocompression systems (Šarevski, 2012a, 2012b) The two phase ejector flow analyses is based on the assumption that saturated vapor-liquid mixture is in thermodynamic equilibrium state at any cross-section of the ejector, and that liquid and vapor are uniformly mixed and flow at the same velocity without inter-phase slip. In the ejector primary nozzle motive (Figure 1) fluid accelerates and expands from the high pressure $p_1$ to the pressure $p_2$ which is lower than centrifugal compressor outlet pressure. The flow at the outlet of the primary nozzle is supersonic, and the nozzle profile is convergent-divergent. At primary nozzle throat cross-section the pressure is equal to the critical and the velocity is equal to the sound velocity $a = \sqrt{\partial p / \partial \rho}$ . Viewing from the existing literature there is a lack of sound velocity data in two phase flow (Wang and Zhang, 2011). The calculation of the sound velocity and the flow analyses of the two phase ejectors are complex tasks. The evaluation of the sound velocity and fluid flow analyses can be carried out by numerical methods ($a = \sqrt{(\Delta p / \Delta \rho)_{s=const}}$ ).

The primary nozzle outlet velocity is

$$c_2 = \Psi_{pr} c_{2a} = [2(h_1 - h_2)]^{1/2} = (2\Delta h_a \eta_{pr})^{1/2} \tag{6}$$

The ejector primary nozzle profile can be obtained applying the energy and continuity equations

$$M = V\rho = Ac\rho = const.; \quad c = \sqrt{2\Delta h}; \qquad A = Mv/c = Mf; \qquad f = v/c;$$

If isentropic change of state is assumed ($s=const.$), then dryness $x$, specific volume $v$ and density ρ are

$$s = xs'' + (1-x)s' = const.; \qquad x = (s - s')/(s'' - s'); \qquad v = xv'' + (1-x)v';$$

The inlet state of the water is: $p=p_1$ ; $t=t_1$ ; $x=0$; $h=h_1$ ; $s=s_1$. The calculated procedure for primary nozzle cross-section area is: $p=p-\Delta p \Rightarrow t \Rightarrow \Delta h \Rightarrow c \Rightarrow x \Rightarrow v \Rightarrow \rho \Rightarrow \Delta\rho \Rightarrow a \Rightarrow f \Rightarrow A$.

The primary nozzle throat cross-section $A=A_{min}$ , ($f=f_{min}$), is critical cross-section, the pressure is critical pressure $p=p_{cr}$ , and the velocity is equal to the sound velocity $c=a$ .

The angle of converging section is $30^o$ and the angle of diverging section is $2^o$ for two phase ejector primary nozzle in the work presented by Banasiac *et al*, (2011). In the work presented by Karwacki *et al*, (2011), the nozzle converging section is profiled, and the angle of diverging section is $8^o$. In this work is suggested that the converging section and the diverging section for two phase ejector primary nozzle to be profiled according to the previous given procedure, assuming constant pressure decrement gradient or constant velocity increment gradient.

This primary flow draws and entrains the secondary flow in the mixing chamber. The secondary flow comes from the centrifugal stage, through the secondary nozzle where it expands. The velocity $c_4$ is

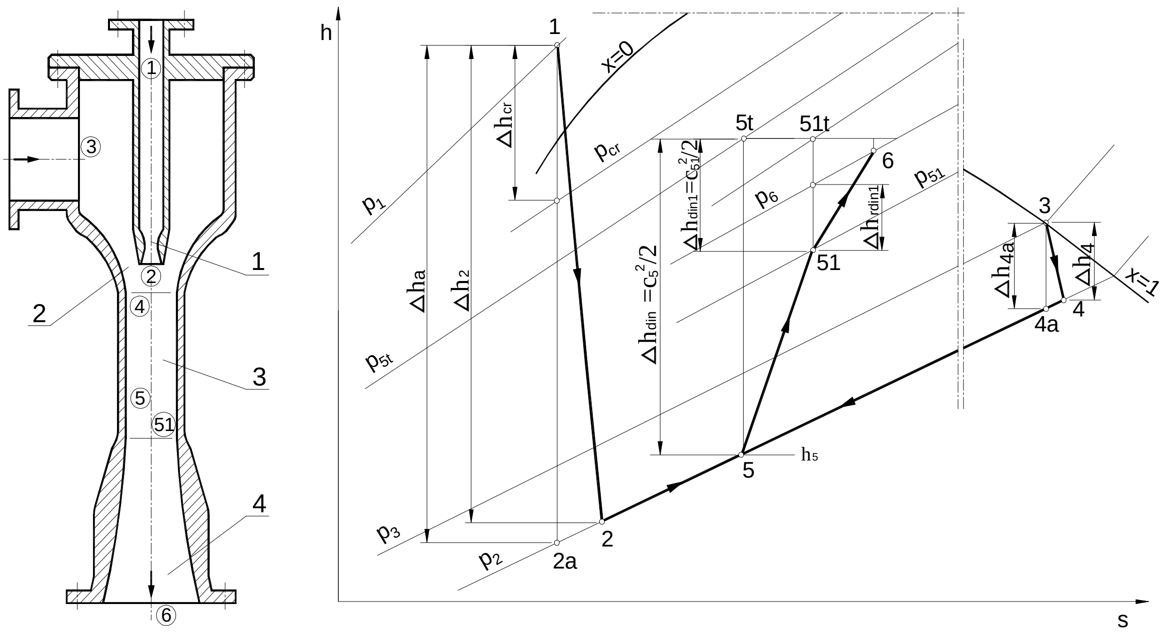


**Figure 1**. Scheme of an ejector and *h* – *s* diagram

$$c_4 = \Psi_{sec} c_{4a} = [2(h_3 - h_4)]^{1/2} = (2\Delta h_{4a}\eta_{sec})^{1/2} \tag{7}$$

The secondary nozzle is formed by the outside profile of the primary nozzle and inside profile of the secondary nozzle. The profiling procedure of the secondary nozzle is similar to that of the primary nozzle. The combined flows are mixed in the mixing chamber (2-5, 4-5), where appears a complex process of momentum transfer. By using the momentum equation for the mixing chamber,

$$p_2 A_2 + p_4 A_4 + M_{pr} c_2 + M_{sec} c_4 = p_5 A_5 + (M_{pr} + M_{sec}) c_5 + P_{fr} \tag{8}$$

Assuming that the cross section areas are $A_2+A_4=A_5$, pressures $p_2=p_4=p_5$ and if the friction forces $P_{fr}$ are comprised with mixing chamber efficiency coefficient $\eta_{mc}$=0.95–0.98, the velocity of the combined flow is

$$c_5 = \eta_{mc}(c_2 m_{pr} + c_4 m_{sec}); m_{pr} = M_{pr}/(M_{pr} + M_{sec}); m_{sec} = M_{sec}/(M_{pr} + M_{sec}); \tag{9}$$

The main losses in the ejector occur in the mixing chamber in the process of momentum transfer. The loss of the kinetic energy or loss of the total pressure is

$$\Delta E = E_1 + E_2 - E_5 = M_{pr} c_2^2/2 + M_{sec} c_4^2/2 - (M_{pr} + M_{sec}) c_5^2/2 = 0{,}5 M_{pr} M_{sec}(c_2^2 - c_4^2)/(M_{pr} + M_{sec})$$
$$\delta_e = \Delta E / E_1 = m_{sec}(c_2^2 - c_4^2)/c_2^2 = (1 - m_{pr})(c_2^2 - c_4^2)/c_2^2 \tag{10}$$

However, in these ejector systems primary mass flow rate is much larger than secondary mass flow rate ($m_{pr} >> m_{sec}$). Therefore this loss of total pressure is negligible. This is significant condition for achievement of high energy efficiency by these thermocompression ejector systems.
The enthalpy of the combined flow is obtained by using the energy equation of the mixing chamber,

$$h_5 = m_{pr} h_2 + m_{sec} h_4 + m_{pr} c_2^2/2 + m_{sec} c_4^2/2 - c_5^2/2 \tag{11}$$

The optimal length of the mixing chamber depends on the ejector operating conditions. According to theoretical and experimental investigations and experience the optimum length is $L_{mc}$=(8–10)$D_{mc}$. The compression of the fluid is achieved as the combine stream flows through the diffuser. The kinetic energy $\Delta h_{din}=c_5^2/2$ in the diffuser is transformed to enthalpy rise, expressed by rise of the pressure, according to the Law of Energy Conservation. The combined flow at the mixing chamber outlet often is supersonic. If the velocity of the combined flow is supersonic then a normal shock wave occurs. The shock wave is a process where sudden change in the flow space appears, the velocity suddenly falls from supersonic to subsonic and the pressure rises.

In two phase flow this complex process is accompanied by mass transfer from one phase to the other. Mach number of the supersonic flow, upstream of the shock wave is $\lambda_1=c_5/a_5>1$. Mach number downstream of the shock wave is $\lambda_2=c_{51}/a_{51}<1$. Across the shock wave $\lambda_1\lambda_2=1$. The sound velocity $a_5$ and $a_{51}$ can be estimated by numerical method. Using the conditional isentropic exponent (Karwacki *et al*, 2011) and/or numerical estimation of the conditional isentropic exponent and according to the gas dynamic theory the pressure rise across the shock wave can be approximately estimated

$$\frac{p_{51}}{p_5}=\frac{\lambda_1^2-(\kappa-1)/(\kappa+1)}{1-(\kappa-1)\lambda_1^2/(\kappa+1)} \tag{12}$$

In the shock wave partially the compression is realized. However, the shock wave is thermodynamic irreversible process, with entropy rise. Additional compression is realized in the subsonic diffuser. According to wide range of publications about subsonic diffuser hydraulic losses, the values of diffuser efficiency $\eta_d$ are from 0.60 up to 0.80, depending on shape and operating conditions. According to the recent experimental investigation of two phase ejectors (Elbel and Hrnjak, 2008; Banasiak *et al*, 2011) the diffuser angle of divergence is $3^o$ –$5^o$. The optimal diffuser angle of divergence for these systems is expected to be lower because the amount of liquid in the mixture is much larger.

## 4. COMPACT R718 WATER CHILLER WITH SINGLE STAGE CENTRIFUGAL COMPRESSOR AND TWO-PHASE EJECTORS

A scheme of a R718 refrigerating system with single stage centrifugal compressor and two phase ejector as a second stage compression device, and *T–s* and *p–h* diagrams of the processes are given in Figure 2 (Šarevski, 2012c). The vapor from the first centrifugal stage directly comes into the second two phase ejector stage, where complex thermal and flow phenomena connected with additional compression, desuperheating and condensation inside the two-phase ejector flow field appears. Direct connection of centrifugal stage and two phase ejector is advantageous for compact centrifugal compressor – two-phase ejector refrigerating unit.

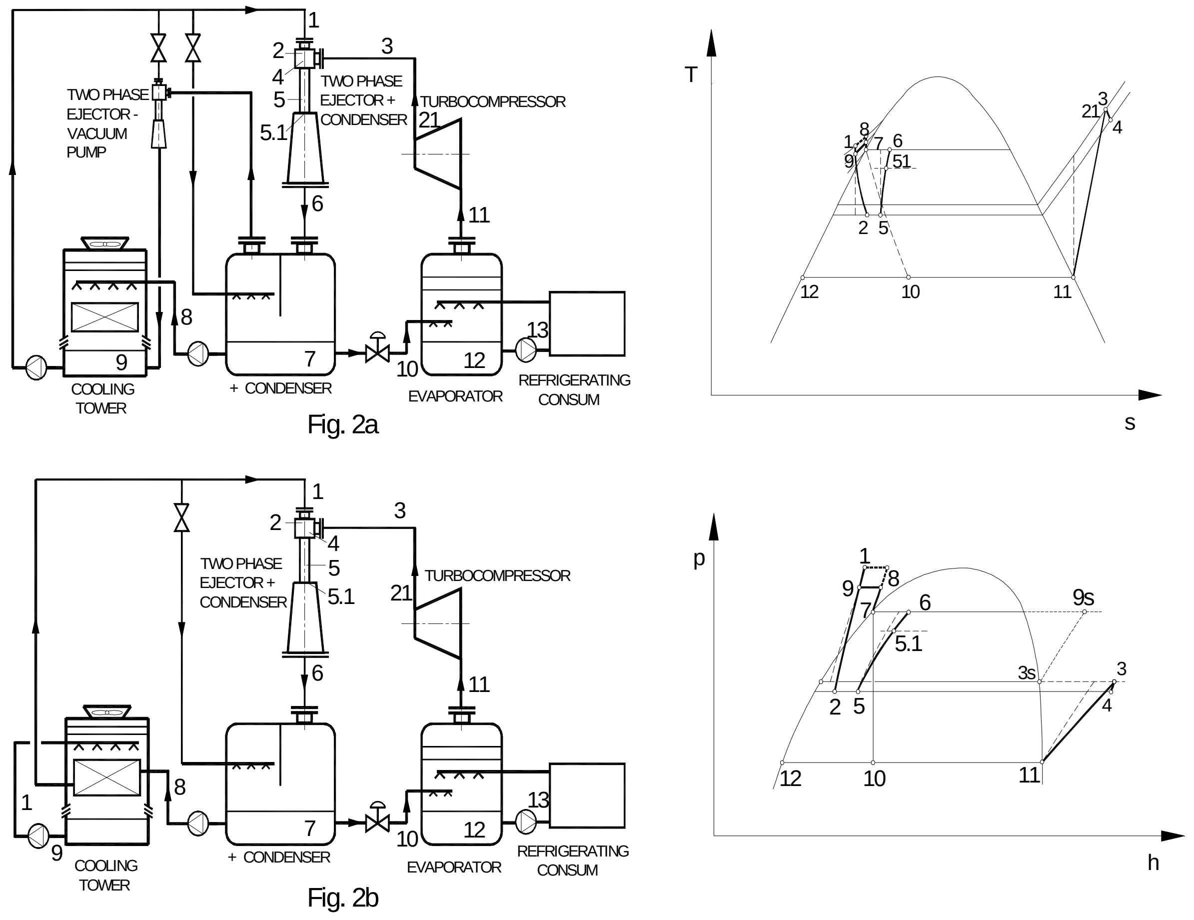


**Figure 2.** Scheme of a R718 refrigerating system with single stage centrifugal compressor and two phase ejector and *T–s* and *p–h* diagrams of the processes

A scheme of a novel compact R718 water chiller with integration of a centrifugal compressor stage and two phase ejectors as a second stage compression device is presented in Figure 3.

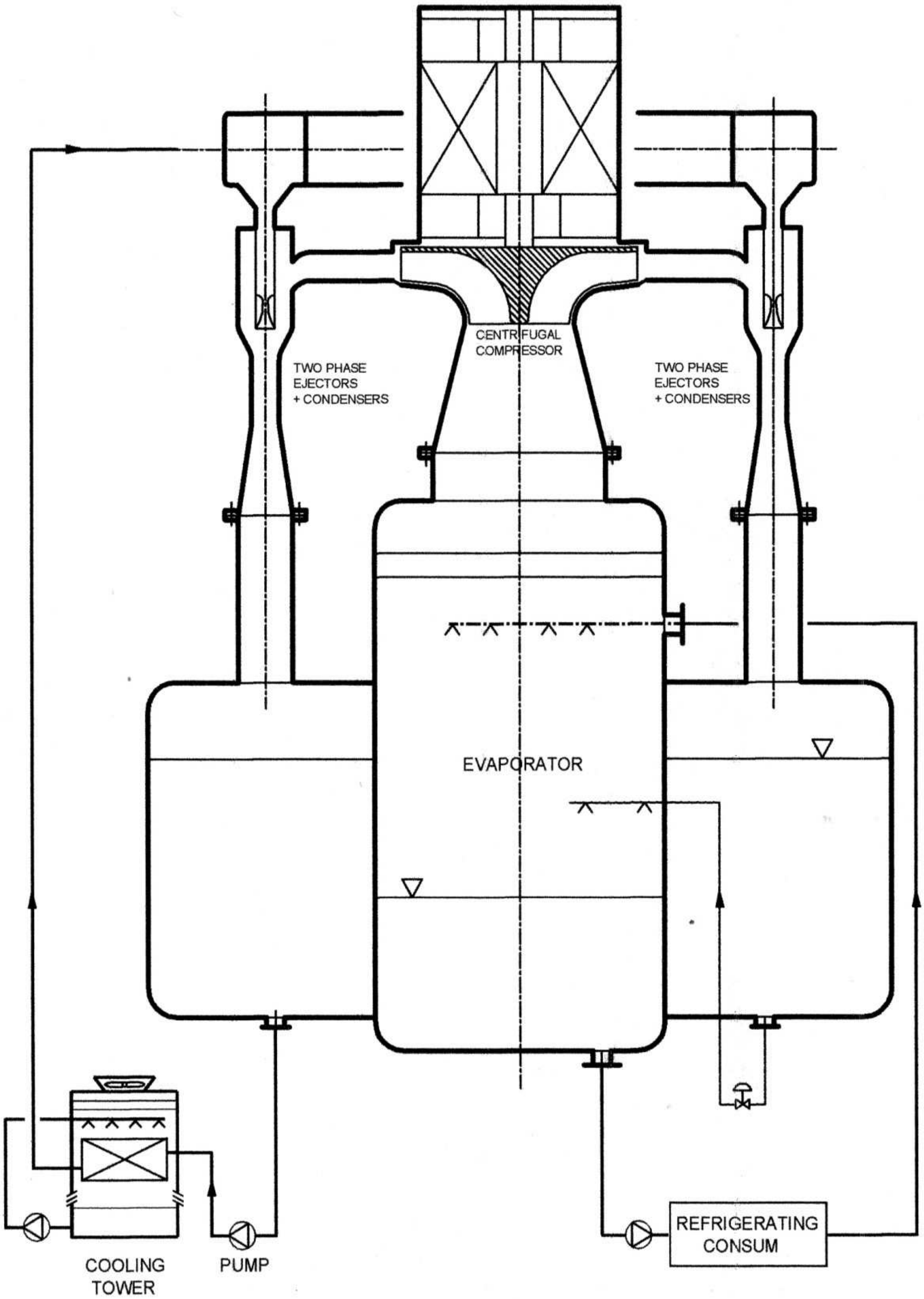


**Figure 3**. Scheme of a novel compact R718 water chiller with centrifugal compressor and two phase ejectors

The evaporator and condenser are with direct evaporation and condensation, without surface heat exchangers. Cooling towers can be direct (Figure 2a) or indirect (Figure 2b). In the direct cooling towers (Figure 2a) water (R718) is exposed directly to the atmospheric contamination with non-condensable gases, solid particles and liquids. That causes additional costs associated with degasifying and cleaning. The vacuum in the system is maintained by a small two phase water ejector vacuum pump, which played a supporting role, working intermittently for pumping a small amount of non-condensable gases. In the indirect cooling towers (Figure. 2b) water (R718) circulates through cooling tower heat exchanger, where heat transfer is still enhanced by wetting the outside of the heat exchanger and utilizing evaporative cooling effect.

The COP of R718 refrigerating system with single stage centrifugal compressor and two phase ejector is

$$COP = q_e \,/\, (\, l_{cc} + l_{ej}\,) \tag{13}$$

$$q_e = h_{11} - h_{10}\,; \quad l_{cc} = h_{21} - h_{11}\,; \quad l_{cc} = l_{cca}/\eta_{cca}\,; \quad \eta_{cca} = 0.7\text{–}0.8\,; \quad l_{ej} = (m_{pr}/m_{sec})(\Delta p_{pr}/(\rho_l \eta_{pump}))\,;$$

$$P_{pump} = \boldsymbol{M}_{pr}\Delta h_{pr}/\eta_{pump} = \boldsymbol{M}_{pr}\Delta p_{pr}/(\rho_l \eta_{pump})\,; \quad m_{sec}\,(q_e + l_1 + l_2) = m_{pr}\, c_l\, \Delta t_l$$

Numerical experiments have been realized for various evaporating and condensing temperatures and various pump characteristics ($\boldsymbol{M}_{pr}, \Delta p_{pr}, \eta_{pump}$), according to the previously explained calculating procedure. The sound velocity is calculated numerically, as well as the isentropic exponent and the profile of the primary nozzle. The calculations are performed for: compressor isentropic efficiency $\eta_{cca}$=0.7–0.8 ($\eta_p$=0.72–0.82); pump efficiency $\eta_{pump}$=0.8; primary nozzle efficiency $\eta_{pr}$=0.9–0.95; secondary nozzle efficiency $\eta_{sec}$=0.9–0.95; mixing chamber mechanical efficiency coefficient $\eta_{mc}$=0.95–0.98; diffuser efficiency $\eta_d$ =0.60–0.80. The estimations of the characteristics of the two phase ejector systems show the following main features: the flow at the outlet of the primary nozzle is supersonic, the nozzle profile is convergent-divergent; primary mass flow rate is much larger than secondary mass flow rate ($m_{pr} >> m_{sec}$), the loss of total pressure in the mixing chamber is negligible; the compression is realized in the shock wave, the shock wave is thermodynamic irreversible process with loss and entropy rise; additional compression is realized in the subsonic diffuser; total efficiency of the two phase ejector ($\eta_{ej}$) defined as a ratio between isentropic compression power from the inter stage pressure (3s, Figure 2) to the condensing pressure (point 9s) and pump power consumption is obtained to be in the range $\eta_{ej}$=0.30–0.53, according to the results of numerical experiments and previously explained calculating procedure. The efficiency of two phase ejector ($\eta_{ej}$) depends on the efficiency of the ejector flow parts, hydraulic pump characteristics, temperature lift $\Delta t = (t_c - t_e)$ ($\Delta t$=(15–5) $^o$C), and subcooling water temperature at the ejector primary nozzle $\Delta t_1 = (t_c - t_1)$ ($\Delta t_1$=(0–7) $^o$C).

The estimated characteristics of the R718 refrigerating system with single stage centrifugal compressor and two phase ejector as results of numerical experiments are given Table 1 for the following conditions: $t_e$=10 $^o$C; $t_c$=35 $^o$C; total pressure ratio Π=4.58; inter stage saturation temperature $t'$($^o$C); centrifugal stage pressure ratio $\Pi_{cc}$; ejector stage pressure ratio $\Pi_{ej}$; water cooling tower temperature 35($^o$C)/$t_1$ ($^o$C); specific centrifugal stage work $l_{cc}$ (kJ kg$^{-1}$); specific ejector work $l_{ej}$ (kJ kg$^{-1}$);
a) for higher efficiency of ejector elements: $\eta_{pr}$=0.95, $\eta_{sec}$=0.95, $\eta_{mc}$=0.98, $\eta_d$ =0.80 and for $\eta_{cca}$=0.8;
b) for lower efficiency of ejector elements: $\eta_{pr}$=0.90, $\eta_{sec}$=0.90, $\eta_{mc}$=0.95, $\eta_d$ =0.60 and for $\eta_{cca}$=0.7.

Table 1. Thermal characteristics of centrifugal compressor / two-phase ejector refrigerating cycle

| | $t'$ $^o$C | $\Pi_{cc}$ | $\Pi_{ej}$ | $t_1$ $^o$C | $\eta_{cca}$ | $\eta_{ej}$ | $l_{cc}$ kJ kg$^{-1}$ | $l_{ej}$ kJ kg$^{-1}$ | *COP* | $t'$ $^o$C | $\Pi_{cc}$ | $\Pi_{ej}$ | $t_1$ $^o$C | $\eta_{cca}$ | $\eta_{ej}$ | $l_{cc}$ kJ kg$^{-1}$ | $l_{ej}$ kJ kg$^{-1}$ | *COP* |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| a) | 22 | 2.15 | 2.13 | 28 | 0.8 | 0.47 | 131 | 227 | 6.60 | 26 | 2.74 | 1.67 | 30 | 0.8 | 0.50 | 175 | 142 | 7.46 |
| b) | | | | | 0.7 | 0.36 | 150 | 288 | 5.40 | | | | | 0.7 | 0.39 | 200 | 181 | 6.22 |
| a) | 23 | 2.29 | 2.00 | 29 | 0.8 | 0.47 | 142 | 208 | 6.77 | 27 | 2.90 | 1.58 | 30 | 0.8 | 0.51 | 186 | 122 | 7.70 |
| b) | | | | | 0.7 | 0.37 | 162 | 262 | 5.59 | | | | | 0.7 | 0.40 | 213 | 156 | 6.43 |
| a) | 24 | 2.43 | 1.88 | 29 | 0.8 | 0.48 | 153 | 185 | 7.01 | 28 | 3.08 | 1.48 | 31 | 0.8 | 0.52 | 194 | 101 | 8.00 |
| b) | | | | | 0.7 | 0.37 | 174 | 240 | 5.72 | | | | | 0.7 | 0.40 | 222 | 131 | 6.70 |
| a) | 25 | 2.58 | 1.78 | 30 | 0.8 | 0.49 | 164 | 163 | 7.25 | 29 | 3.26 | 1.40 | 31 | 0.8 | 0.53 | 207 | 90 | 8.30 |
| b) | | | | | 0.7 | 0.38 | 187 | 210 | 6.00 | | | | | 0.7 | 0.41 | 237 | 116 | 7.00 |

The estimated *COP* of the refrigerating system in Figure 2 and Figure 3 is in range *COP*=(5.4–8.3), depending on the efficiency of the ejector flow parts, compressor efficiency and hydraulic pump characteristics. This *COP* comprises all power consumption, including condenser-cooling tower subsystem. Therefore the total energy efficiency of these refrigerating systems may also be increased.

The application of the compact R718 water chiller with centrifugal compressor and two phase ejectors concerning temperate lift is estimated to be in the range $\Delta t=(t_c–t_e)$=(20–35) K, which approximately correspond to total pressure ratio $\Pi=p_c/p_e \approx 3.0–8.0$. This range covers wide application in the refrigeration and air conditioning.

It is necessary to conduct further experimental investigations to obtain optimal two phase ejector flow field, as well as optimal performance characteristics of the motive pump. The two phase ejector calculating procedure will be improved with obtained experimental data.

## 5. CONCLUSIONS

A novel compact R718 water chiller with centrifugal compressor and two phase ejectors as a second stage compression device is proposed and described. The limitations of the R718 centrifugal compressor stage pressure ratio are discussed, and possibilities for introduction of two-phase ejectors in the R718 refrigerating systems with direct evaporation and condensation are analyzed. The complex thermal and flow phenomena connected with additional compression, desuperheating and condensation inside the two-phase ejector flow field are investigated and performance characteristics are estimated.

Numerical experiments have been realized for various evaporating and condensing temperatures and various motion pump characteristics. The estimations of the characteristics of the two phase ejector systems show the following main features: the flow at the outlet of the primary nozzle is supersonic, the nozzle profile is convergent-divergent; primary mass flow rate is much larger than secondary mass flow rate ($m_{pr}$>>$m_{sec}$), the loss of total pressure in the mixing chamber is negligible; the compression is realized in the shock wave, the shock wave is thermodynamic irreversible process with loss and entropy rise; additional compression is realized in the subsonic diffuser; total efficiency of the two phase ejector defined as a ratio between isentropic compression power from the inter stage pressure to the condensing pressure and pump power consumption is estimated to be in the range ($\eta_{ej}$=0.30–0.53), depending on the efficiency of the ejector flow parts, hydraulic pump characteristics, thermotransforming temperature lift $\Delta t=(t_c–t_e)$, ($\Delta t$=(15–5) $^oC$), and subcooling water temperature at the ejector primary nozzle $\Delta t_1=(t_c–t_1)$ ($\Delta t_1$=(0–7) $^oC$).

The estimated *COP* of this refrigerating system is in range *COP*=(5.4–8.3), depending on the efficiency of the ejector flow field and hydraulic pump characteristics. This *COP* comprises all power consumption, including condenser-cooling tower subsystem. Therefore the total energy efficiency of these refrigerating systems may also be increased.

The application of these refrigerating systems concerning to the temperate lift is assumed to be in the range $\Delta t=(t_c–t_e)$=(20–35)(K), which approximately correspond to total pressure ratio $\Pi=p_c/p_e \approx$ 3.0–8.0. This range covered wide application in the refrigeration and air conditioning. Wide range of capacities of R718 single stage centrifugal compressor – two-phase ejector refrigerating unites (from tens kW up to (1–2) MW) can be obtained. The implementation of the two-phase ejector in the R718 centrifugal compressor refrigerating systems causes simplification of the R718 unit, and reduction of their size and cost.

## NOMENCLATURE

| | |
|---|---|
| $a$ | Speed of sound (m s$^{-1}$) |
| $A$ | Flow cross-section (m$^2$) |
| $b$ | Impeller width (m), $\bar{b}_2 = b_2 / D_2$ |
| $c$ | Velocity (m s$^{-1}$) |
| $\boldsymbol{c}$ | Specific compressor displacement (m$^3$ J$^{-1}$) |
| $c_l$ | Water specific heat capacity (J kg$^{-1}$ K$^{-1}$) |
| $c_m$ | Meridional velocity component (m s$^{-1}$) |
| $c_u$ | Peripheral velocity component (m s$^{-1}$) |
| $COP$ | Coefficient of performance |
| $D$ | Diameter (m) |
| $h$ | Enthalpy (J kg$^{-1}$) |
| $k_v$ | Density ratio |
| $k_1$,$k_2$ | Slip factor coefficients, |
| $l$ | Compressor work (J kg$^{-1}$) |
| $l_1$ | Centrifugal compressor work (J kg$^{-1}$) |
| $l_2$ | Ejector compression work (J kg$^{-1}$) |
| $M$ | Mach number |
| $M_u$ | Peripheral Mach number |
| $\boldsymbol{M}$, $M$ | Mass flow rate (kg s$^{-1}$) |
| $m$ | Relative mass flow rate |
| $n$ | Speed of rotation (rev s$^{-1}$) |
| $P_{pump}$ | Pump power consumption (W) |
| $P_{fr}$ | Friction forces (N) |
| $p$ | Pressure (Pa, bar) |
| $Q_e$ | Cooling capacity (W) |
| $q_v$ | Volumetric cooling capacity (J m$^{-3}$) |
| $q_e$ | Specific refrigerating effect (J kg$^{-1}$) |
| $R$ | Gas constant (J kg$^{-1}$ K$^{-1}$) |
| $R_\mu$ | Universal gas constant (J kmol$^{-1}$ K$^{-1}$) |
| $r$ | Coefficient of impeller reactivity, |
| $s$ | Specific entropy (J kg$^{-1}$ K$^{-1}$) |
| $T$, $t$ | Temperature (K, $^o$C) |
| $u_2$ | Impeller peripheral speed (m s$^{-1}$) |
| $V$ | Volumetric flow rate (m$^3$ s$^{-1}$); Compressor capacity (m$^3$ s$^{-1}$) |
| $v$ | Specific volume (m$^3$ kg$^{-1}$) |
| $x$ | Dryness (quality) |

Greek letters

| | |
|---|---|
| $\beta_2$ | Impeller blade outlet angle ($^o$) |
| $\Delta p_{pr}$ | Pump pressure rise (Pa) |
| $\Delta t$ | Temperature difference ($^o$C, K) |
| $\Delta t_l$ | Water temperature drop in cooling tower (K) |
| $\eta$ | Efficiency |
| $\kappa$ | Isentropic exponent |
| $\mu$ | Molecular mass (kg kmol$^{-1}$) |
| $\mu_s$ | Sleep factor |
| $\xi$ | Compressibility factor |
| $\Pi$ | Pressure ratio |
| $\rho_l$ | Water mean density |
| $\Psi$ | Velocity coefficient |
| $\psi$ | Work coefficient |
| $\psi_t$ | Theoretical work coefficient |
| $\varphi$ | Flow rate coefficient |
| $\varphi_{2d}$ | Flow rate coefficient at design point |
| $\tau$ | Contraction of flow cross-section |

Subscripts

| | |
|---|---|
| $a$ | Isentropic |
| $c$ | Condensation |
| ***cc*** | Compressor |
| $cr$ | Critical |
| $e$ | Evaporation |
| ***ej*** | Ejector |
| $h$ | Gas dynamic |
| $mc$ | Mixing chamber |
| $p$ | Polytropic |
| $pr$ | Primary |
| $sec$ | Secondary |
| 0 | Compressor inlet |
| 1 | Impeller cascade inlet |
| 2 | Impeller outlet |
| 1– 13; 21; 51 | States in Figure 1 and Figure 2 |
| ’ | Saturated liquid |
| ” | Saturated vapor |

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